COMHPTES

Flexible Compact Modular Heat Pump and PCM based Thermal Energy Storage System for Heat and Cold Industrial Applications

D2.1 Definition of high temperature heat pump cycles and boundary conditions

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Author: Jan Oledzki, Jerémie Lagarde

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Executive summary

Task 2 will define the preliminary design of the innovative high temperature heat pump architecture (cycle configurations and components) including the main integration boundaries for the key technologies that will be developed (compressor/expander and heat exchangers). Within this, deliverable 2.1 proposes the boundary conditions for the design of the power-limited lab-scale heat pump test rig system. High temperature drying, simultaneous hot & cold and steam generation are considered for defining maximum and minimum operating conditions of the heat pump part of the COMHPTES project. It is proposed that the high pressure circuit should have a maximum operating pressure of 200 barG and operating temperature of 315°C. For the low pressure circuit, an operating pressure of 50 barG is recommended with a maximum operating temperature of 140°C and minimum of -20°C. Including a recuperator should not be necessary for the COMHPTES demonstrator if steam with no solar source is discarded as an application (steam with solar is still an option). The effect of heat exchanger pinch temperature, pressure loss and design pressure on COP is studied showing that increasing pinch temperature will have the biggest negative impact on system COP. Reducing design pressure should be avoided due to the negative impact on COP. The power limitations imposed by the test rig will result in lower efficiencies than would be expected form a “real” system deployed in the field.

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1. Introduction

1.1 Objectives

The primary aim of Task 2.1 is to define the boundary conditions for the COMHPTES high temperature heat pump (HTHP) and steer equipment design such that all components are compatible and able to meet the objectives of the wider project. This study aims to propose a reasonable limit for pressures, temperatures and flowrates of the refrigerant within the heat pump cycle which will inform the design of the compressor/expander, heat exchangers, piping and other auxiliary equipment such as valves, instrumentation, pressure vessels and the hot and cold heat transfer loops for simulating thermal loads.

1.2 Representative Applications

As with many large heat pump projects, the design and integration of the system is application dependant, thus a critical first step is to identify some case studies that line up with the objectives of the COMHPTES project – i.e., industrial end-users with a heat demand from 0.5 to 10 MW_{th} and temperature requirements less than 225-300°C.

Due to a lack of end-user applications at the time of writing, the following general processes were identified as ones of interest to consider for boundary conditions within the COMHPTES project.

1.2.1 High Temperature Drying

High temperature drying (referred to from now on as drying) is the process of removing moisture from a product, in this context, usually to produce powder or other homogenous dried substance. In the food industry, various dryer types are used to produce products such as milk powder, dried whey, dried sourdough starters and instant coffee. Various drying process are also employed in the manufacturing industry for producing bulk powders and fertilisers. Table 1, below, summarises some typical drying processes and common temperatures used.

Table 1: Drying processes and typical temperatures[1], [2]

Process	Temperature
Whey and Milk Powder	150-300°C
Coffee	270°C
Fertilisers	200-400°C
Extruded Feed	100-175°C
Charcoal Briquettes	130-170°C

The drying process typically reduces the temperature of the hot inlet air to the range of 50-100°C due to the evaporative cooling effect, resulting in a good waste heat source. Furthermore, in many industries, especially ones handling food, drying air cannot be recycled due to hygiene and contamination concerns. Fresh air must be taken in and heated up to drying temperatures, increasing the temperature glide.

An example drying process which may be used for defining boundary conditions is summarised in Table 2. The heating duty is not considered as this will only influence scale and not the cycle temperatures. Figure 1 shows a diagram of what a dryer with a heat pump may look like.

Table 2: Example drying process for defining HPHT boundary conditions.

Parameter	Temperature
Ambient, Fresh Air	15°C
Hot, Dry Air	210°C
Moist Air	85°C
Cooled Exhaust Air	<30°C

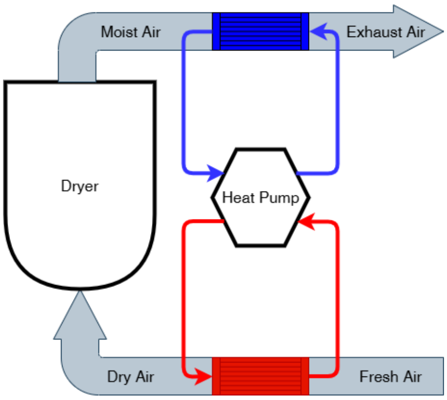

Figure 1: Diagram of spray dryer with heat pump.

1.2.2 Steam

Approximately 40% of EU process heat demand is met by steam (with another 50% of heat above 400°C)[3]. Consequently, a large portion of processes in the temperature range of interest will use steam as the heat transfer medium making it a big focus point for industrial high temperature heat pumps as boiler replacements. Steam at up 10 barG (184°C) will be considered for cycle definition. A subcooling of 10 K is applied to the condensate and 10 K of superheat to the steam. This is summarised in Table 3.

Table 3: Steam conditions for defining HPHT boundary conditions.

Parameter	Value
Steam pressure	10 barG
Saturation temperature	184°C
Superheated steam temperature	194°C
Condensate temperature	174°C
Source inlet temperature	40°C
Source outlet temperature	15°C
Solar source temperature	150°C

1.2.3 Hot and Cold

Although heating and cooling duties are not typically matched and often not in phase in many processes, the use of thermal energy storage allows these to be catered to with a single system. The processing of milk is a good example with production often being batched but requiring a spread of temperatures and demands. Table 4 summarises some common dairy processes and associated temperatures.

Table 4: Typical temperature ranges in dairy processing[1]

Process	Temperature
Fresh Product Storage	0-5°C
Ice-cream etc.	<-20°C
Pasteurisation	70-100°C
Sterilisation / UHT	115-150°C
Cheese Production	50-65°C
Drying (milk, whey)	150-300°C

For the purpose of defining heat pump boundary conditions (within the COMHPTES scope), we can consider the system presented in Table 5.

Table 5: Example dairy process for defining HPHT boundary conditions.

Parameter	Value
Sink process fluid	Water
Sink inlet temperature	70°C
Sink outlet temperature	150°C
Source process fluid	Water-Glycol/ Air
Source inlet temperature	5°C
Source outlet temperature	0°C

2. Modelling Methodology

2.1 Basic Heat Pump Cycle

A simple vapour compression heat pump cycle is used to define the operating conditions of the test rig (Figure 2). It is assumed, that the electrical power of the motor driving the heat pump is limited to 100 kW. This is based on the availability of two 160 kW_e busbars in the lab and an assumption that other equipment is connected to these, including one taken up by heating and cooling equipment for the test rig. The inlet and outlet temperatures for the heat sink and source are defined as per the section above, as well as a thermal duty. The thermodynamic properties for the CO_2 working fluid are calculated using CoolProp's excel plug in[4]. Constants used in the modelling of the compressor-expander are presented in Table 6. It should be noted that a lower value of mechanical efficiency (75%) is used as the lab power limitations result in only a partially populated machine with 3-6 cylinders. A full scale 12-cylinder system would target mechanical efficiencies of up to 92.5%. Pressure drop in the heat exchangers, as a percentage of the inlet pressure, is also applied to the cycle calculation (Table 7).

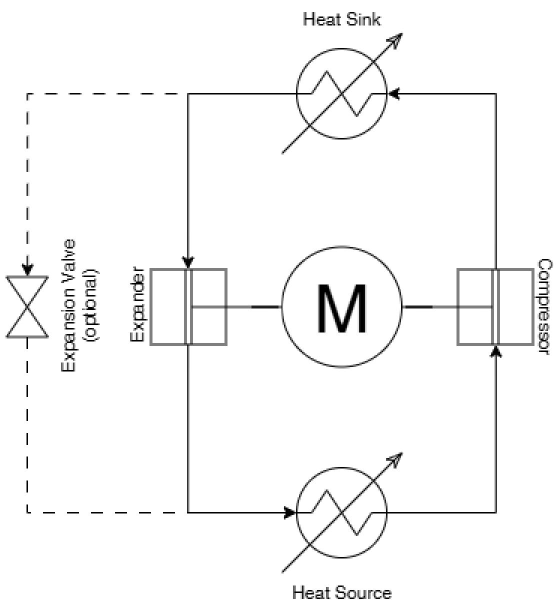

Figure 2: Simple heat pump diagram.

Table 6: SynchroStor's Compressor-Expander machine parameters. Based on testing results and theoretical targets.

Parameter	Unit	Value
Cylinder Volume	cc/rev	432
Shaft speed	rpm	1000
Volumetric efficiency (compression)	%	90
Volumetric efficiency (expansion)	%	80
Isentropic efficiency	%	90
Mechanical efficiency (25% Load)	%	75
Electrical efficiency	%	97.5
Maximum outlet pressure	barA	200
Maximum inlet pressure	barA	50
Maximum inlet temperature	°C	150
Maximum pressure ratio	-	10

The cycle is optimised to maximise the hot coefficient of performance (COP), defined as the heat output at the heat sink over the total work done to drive the compressor. The optimisation algorithm is set to vary heat source pressure (or low pressure), heat sink pressure (or high pressure), compressor inlet temperature and the machine utilisation – defined as the percentage of cylinders being used on the machine. A series of constraints are applied to the optimisation problem to ensure that the solution is feasible, including the maximum and minimum pressure (Table 6) and the pinch temperature in the heat exchangers (Table 7). There is also a maximum pressure ratio limit of 10 and a maximum compressor suction temperature of 150°C limited by the current design of the machine.

Table 7: Baseline parameters used for optimising cycle.

Parameter	Unit	Value
Heat sink minimum pinch temperature	K	20
Heat source minimum pinch temperature	K	20
Heat sink pressure drop	%	1
Heat source pressure drop	%	1

2.2 Heat Exchanger Pinch Temperature

The pinch temperature represents the minimum temperature difference between the hot and cold stream at any point within the heat exchanger. To allow for the thermal oil system (or other fluid) and TES between the heat pump and the process, we can allow for a higher minimum pinch temperature in the sink and source heat exchanger. This essentially simulates the losses from heat exchangers stacking. For example, if one was to allow for 10 K pinch temperature in each heat exchanger, the total pinch temperature would be equivalent to 20 K in the model, not accounting for other losses. Figure 3 shows how the pinch temperature manifests itself in two typical heat transfer arrangements – the pinch is not necessarily at the extremities of the heat exchanger.

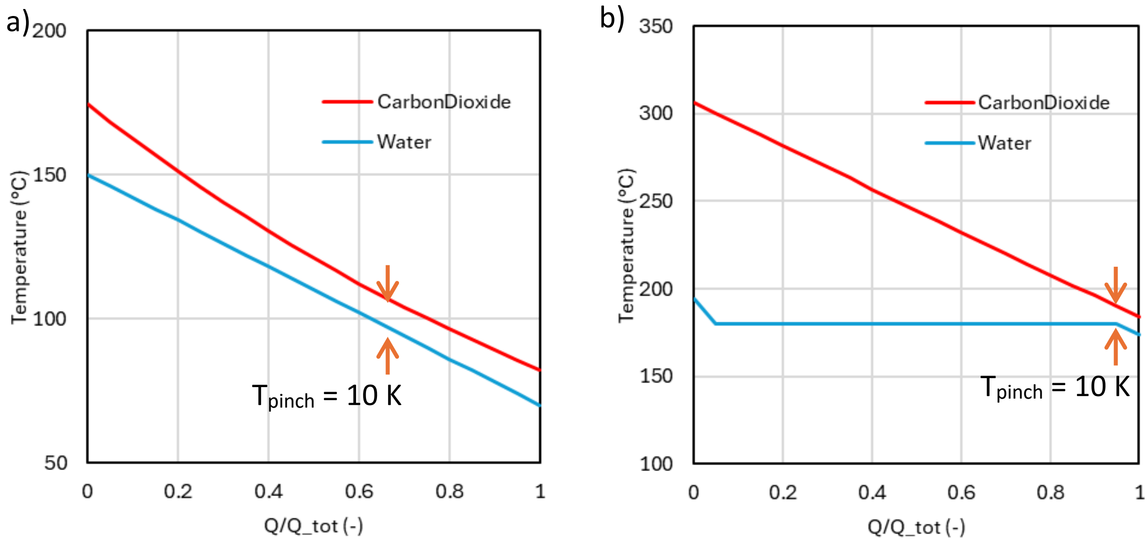

Figure 3: Temperature profile showing pinch point for a) heating water and b) generating steam

2.3 Recuperator

The recuperator or internal heat exchanger (IHX) allows for excess heat to be recovered after the heat sink to preheat the gas entering the compressor, resulting in higher discharge temperatures. This is especially useful for cycles where the outlet of the sink is still at a high temperature (for example, steam generation) enabling a higher COP.

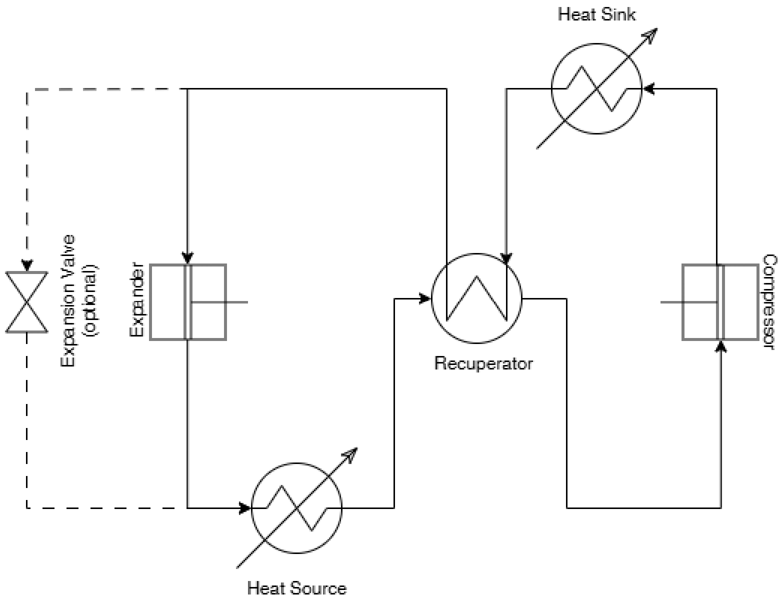

Figure 4: Diagram of a heat pump with a recuperator.

The recuperator has the same constraints imposed on it as other heat exchange processes in the cycle including minimum pinch temperature and pressure drop. Although a recuperator is not planned for the demonstration skid at KTH it is included for the purpose of discussion and for potential inclusion in techno-economic analysis.

3. Boundary Conditions

The results for each of the applications defined in Section 1.2 are presented below for the base constants (Table 7) in a simple heat pump cycle. Thermal power and flowrate can be scaled for higher input power (e.g., x1.6 if electrical power is increased to 160 kW_e) with COP, temperature and pressure remaining constant.

3.1 Drying

The drying application is the best fit for a transcritical CO_2 heat pump due to the large temperature glide in the heat sink and waste heat available at relatively high temperatures. The results from the modelling for the base case are presented in Table 8. Due to the large temperature glide and high temperature of waste heat, the flowrate of the refrigerant is relatively low, and paired with a high suction pressure (high refrigerant density) only two compression cylinders are required for 100 kW of input power. Expansion helps to recover a small amount of energy increasing the COP of the cycle with expansion by approximately 20%.

Table 8: System parameters for drying case study with 100 kW_e input power

Parameter	Unit	Expansion	Throttling
Heat Sink Duty	kW_{th}	190	170
Heat Sink Inlet Temperature	°C	240	259.7
Heat Sink Outlet Temperature	°C	64.7	54.4
Heat Sink Pressure	bara	200	200
Heat Source Duty	kW_{th}	124	97
Heat Source Inlet Temperature	°C	-0.8	-7.6
Heat Source Outlet Temperature	°C	64.7	64.8
Heat Source Pressure	bara	33.8	28.1
COP	-	1.90	1.70
Compression Cylinders (actual)	-	1.6 (2)	1.5 (2)
Expansion Cylinders (actual)	-	0.6 (1)	0
Refrigerant flowrate	kg/s	0.61	0.47
Shaft Work	kW	98	98

Figure 5 gives a visual representation of the two cycle conditions described in Table 8. While the two cycles are not identical, they are very similar with the heat sink pressure at 200 bara in both cases (interestingly constrained to their limit). Only the boost pressure differs slightly (4 bar difference), which as a result affects the evaporation temperature.

Figure 6 shows how the COP of a heat pump for drying changes with sink and source temperature. Logically, the COP increases as the heat source temperature increases and as the dry air temperature decreases (i.e. temperature lift decreases).

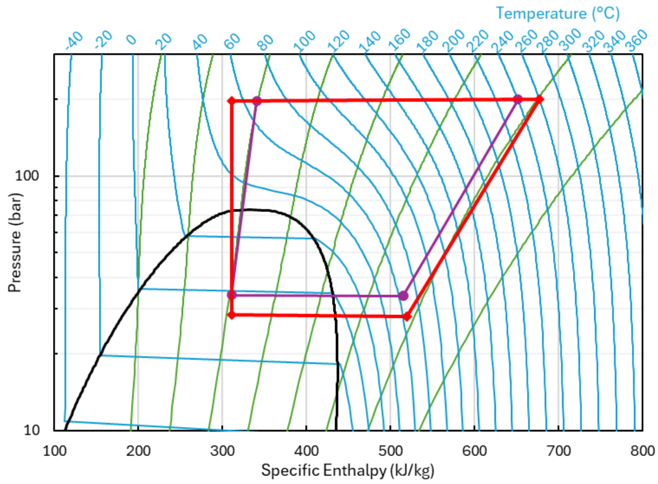

Figure 5: Pressure-Enthalpy (P-h) diagram for CO_2 heat pump showing the drying application using expansion (purple) and throttling (red)

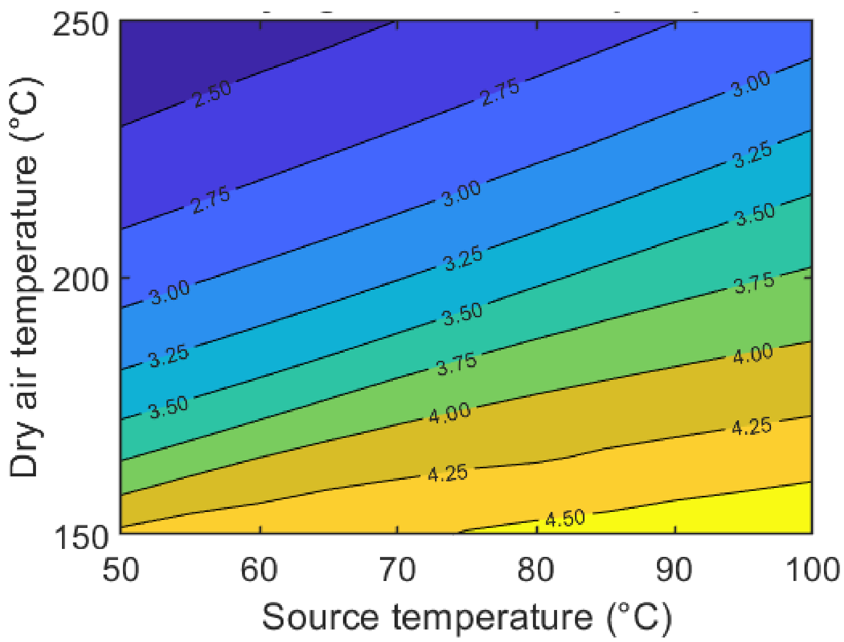

Figure 6: COP map for a simple CO_2 heat pump with expansion for drying with varying source temperature. T_{pinch} = 5 K, η_{mech} = 92.5%

3.2 Steam

The temperature profiles of CO_2 and evaporating water are not well matched (Figure 3(b)) which makes generating steam a difficult application. Even with a recuperated cycle (Figure 4) the COP is limited especially as the steam pressure increases; this is illustrated in Figure 7. A recuperator is essential for recovering the heat left over after steam generation in order to boost the compressor suction temperature, thus reducing the temperature lift. An alternative to using a recuperator is utilising the Absolicon Solar Thermal Collector, which also boosts the compressor suction temperature. Expansion is considered for all steam cases. Generating steam without a recuperator or alternative high temperature heat source is very unlikely to be a feasible electrification strategy.

3.2.1 Steam with Recuperator

Table 9 shows the results for a heat pump with a recuperator. A heat source from waste heat or cooling water is still considered to provide heat to the system. The COP is still quite low and the additional cost of the system due to the recuperator is likely to be higher.

Table 9: System parameters for steam generation with recuperator and expansion with 100 kW$_e$ input power. Heat source assumed to be available at 40°C and cooled down to 25°C.

Parameter	Unit	Value
Heat Sink Duty	kW$_{th}$	94
Heat Sink Inlet Temperature	°C	385.9
Heat Sink Outlet Temperature	°C	194.9
Heat Sink Pressure	bara	200
Heat Source Duty	kW$_{th}$	32
Heat Source Inlet Temperature	°C	-12.1
Heat Source Outlet Temperature	°C	19.7
Heat Source Pressure	bara	24.7
COP	-	0.94
Compression Cylinders (actual)	-	1.9 (2)
Expansion Cylinders (actual)	-	0.9 (1)
Refrigerant flowrate	kg/s	0.39
Shaft work	kW	98
Recuperator Duty	kW$_{th}$	51
Recuperator High Pressure Outlet Temperature	°C	113.2
Recuperator Low Pressure Outlet Temperature	°C	150.0

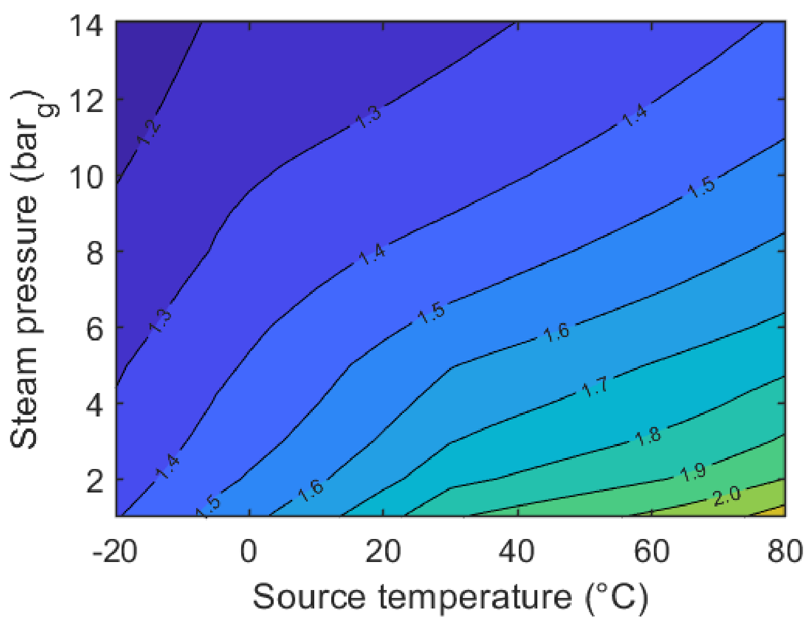

Figure 7: COP map for a recuperated CO_2 heat pump for generating steam at different source temperatures. T_{pinch} = 5 K, η_{mech} = 92.5%

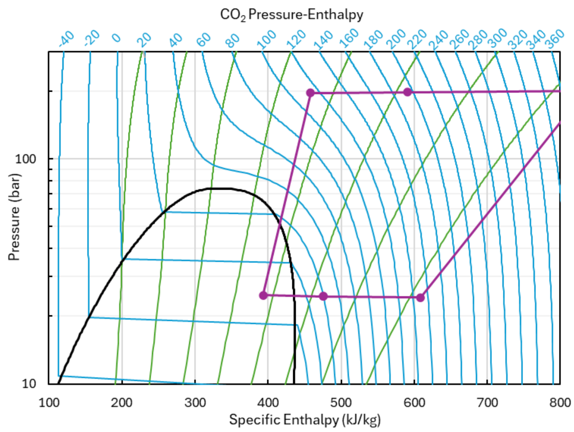

Figure 8: Pressure-enthalpy (P-h) diagram for CO2 heat pump generating steam with recuperator and heat source at 40°C.

3.2.2 Steam with Solar

The heat from the solar collector noticeably improves the COP of the heat pump, presented in Table 10. Figure 9 shows that the cycle is completely in the vapour phase of CO_2 and therefore requires a slightly larger source heat exchanger.

Table 10: System parameters for steam generation with solar thermal collector delivering heat at 150°C and expansion with 100 kW$_e$ input power.

Parameter	Unit	Value
Heat Sink Duty	kW$_{th}$	99
Heat Sink Inlet Temperature	°C	314.2
Heat Sink Outlet Temperature	°C	198.2
Heat Sink Pressure	bara	200
Heat Source Duty	kW$_{th}$	53
Heat Source Inlet Temperature	°C	55.4
Heat Source Outlet Temperature	°C	129.8
Heat Source Pressure	bara	36.3
COP	-	0.99
Compression Cylinders (actual)	-	2.0 (3)
Expansion Cylinders (actual)	-	1.7 (2)
Refrigerant flowrate	kg/s	0.66
Shaft Work	kW	98

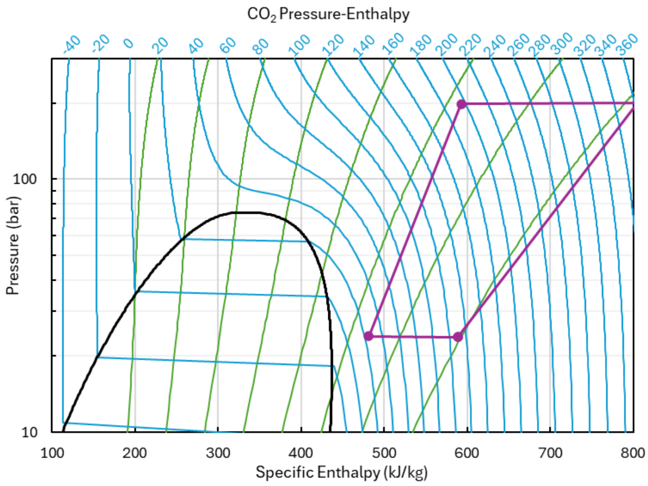

Figure 9: Pressure-enthalpy (P-h) diagram for CO2 heat pump generating steam with heat supply from solar collector.

3.2.3 Steam and Other Thermal Load

In some processes steam is used to generate hot water which is then used to heat the process fluid. This may be the case in systems such as dairy pasteurisation where the steam temperature is too high and may burn the product. Generating additional steam to then use it to heat water is fairly inefficient, especially in a heat pump system. Alternatively, a heat pump could generate steam and then utilise the remaining heat in the working fluid to heat water, this is shown in Figure 10Figure 10. This would increase the glide and boost the COP of the heat pump whilst also translating to operational improvements by reducing the size of the steam system. This should be an area of focus for COMHPTES as it addresses the very large steam market but also leverage some of the advantages of using CO_2 as a working fluid. Table 11 shows the cycle parameters for generating equal heat load of hot water and steam.

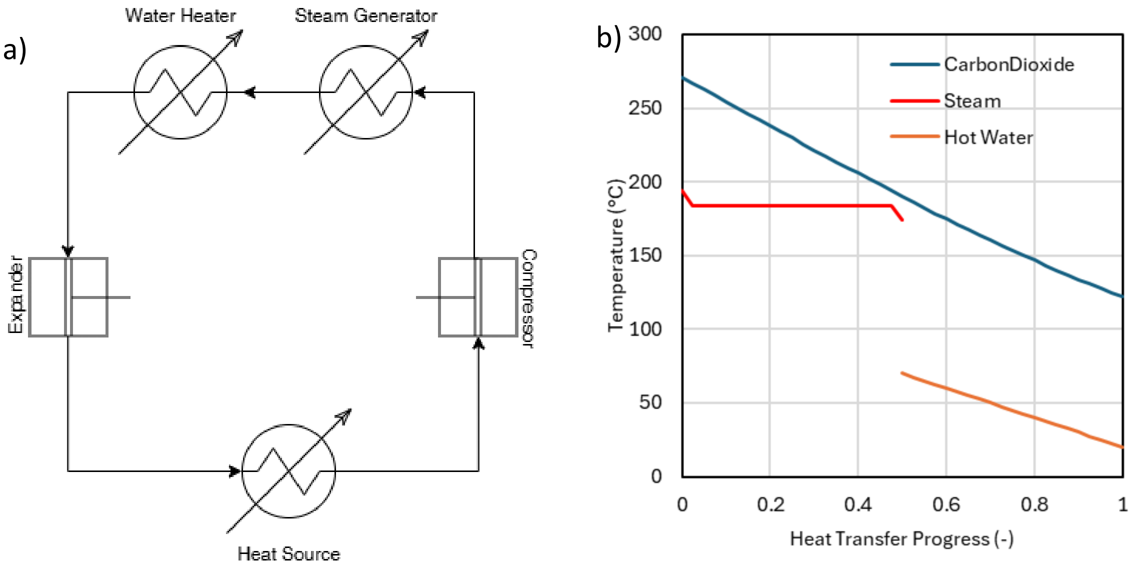

Figure 10: a) Diagram of heat pump generating steam and hot water and b) temperature profile in the two heat sinks for steam at 10 barG and hot water at 70°C.

Table 11: System parameters for steam and hot water generation with solar thermal collector delivering heat at 150°C and expansion with 100 kW$_e$ input power.

Parameter	Unit	Value
Steam Generator Duty	kW$_{th}$	79
Water Heater Duty	kW$_{th}$	79
Steam Generator Inlet Temperature	°C	279.5
Steam Generator Outlet Temperature	°C	199.9
Water Heater Outlet Temperature	°C	132.2
Heat Sink Pressure	bara	200
Heat Source Duty	kW$_{th}$	102
Heat Source Inlet Temperature	°C	23.8
Heat Source Outlet Temperature	°C	129.8
Heat Source Pressure	bara	49.3
COP	-	1.57
Compression Cylinders (actual)	-	1.6 (2)
Expansion Cylinders (actual)	-	1.0 (1)
Refrigerant flowrate	kg/s	0.76
Shaft work	kW	98

3.3 Hot & Cold

By leveraging the ability of a CO_2 heat pump to reach very cold temperatures and utilising that cold to provide process cooling or chilling, the overall COP of the system can be improved. Although the system results in a lower hot COP due to the higher lift required (compared to drawing heat from ambient) some end-users may find it appealing to have both heating and cooling provided by a single system. Table 12 shows that active expansion significantly improves the performance of the system in terms of hot and cold COP, resulting in a combined COP that is 69% higher.

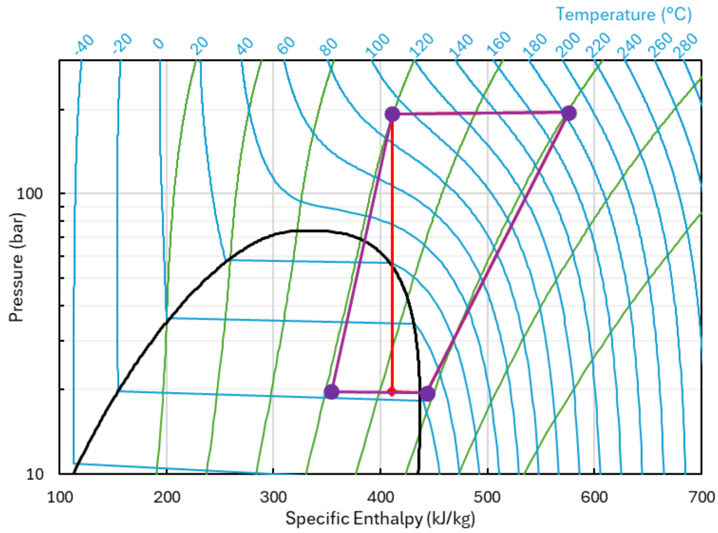

Figure 11: Pressure-Enthalpy (P-h) diagram for CO_2 heat pump showing the hot & cold application using expansion (purple) and throttling (red)

Table 12: System parameters for hot & cold case study with 100 kW_e input power

Parameter	Unit	Expansion	Throttling
Heat Sink Duty	kW_{th}	120	91
Heat Sink Inlet Temperature	°C	183.4	183.4
Heat Sink Outlet Temperature	°C	92.0	92.0
Heat Sink Pressure	bara	195	195
Heat Source Duty	kW_{th}	64	18
Heat Source Inlet Temperature	°C	-20	-20
Heat Source Outlet Temperature	°C	-15.3	-15.3
Heat Source Pressure	bara	19.5	19.5
COP (hot)	-	1.20	0.91
COP (cold)	-	0.64	0.18
COP (total)	-	1.84	1.09
Compression Cylinders (actual)	-	2.3 (3)	1.7 (2)
Expansion Cylinders (actual)	-	1.8 (2)	0
Refrigerant flowrate	kg/s	0.72	0.55
Shaft work	kW	98	98

4. Further Analysis

There are several factors which affect the design of the system in terms of overall efficiency but also the cost of equipment. It is expected, that apart from the compressor/expander, the primary driver of the heat pump cost will be the heat exchangers. In this section, we consider the effect of pressure drop, pinch temperature and design pressure on the heat pump, as these parameters will need to be considered to arrive at a suitable solution in terms of CAPEX and OPEX.

As part of the COMHPTES project a techno-economic model of the system will be developed and, in some ways, this chapter can be seen as a very rough preamble to this analysis. But the real reason for the further analysis is to provide some technical design and budgeting guidance for the heat exchanger design optimization (work package 2.2) and for the test rig design (work package 4.1).

4.1 Theoretical Mechanical Efficiency

SynchroStor's compressor/ expander is designed for 12-cylinders with an approximate drive power of 300-600 kW_e for the considered operating conditions, thus by limiting the input power to 100 kW_e the compressor is operating quite far from its design conditions. This is illustrated in Figure 12, which shows that at a lower machine utilisation (e.g., less than or 3 cylinders instead of 12) will result in lower mechanical efficiencies of around 75% compared to the design target efficiency of up to 92.5% at full load.

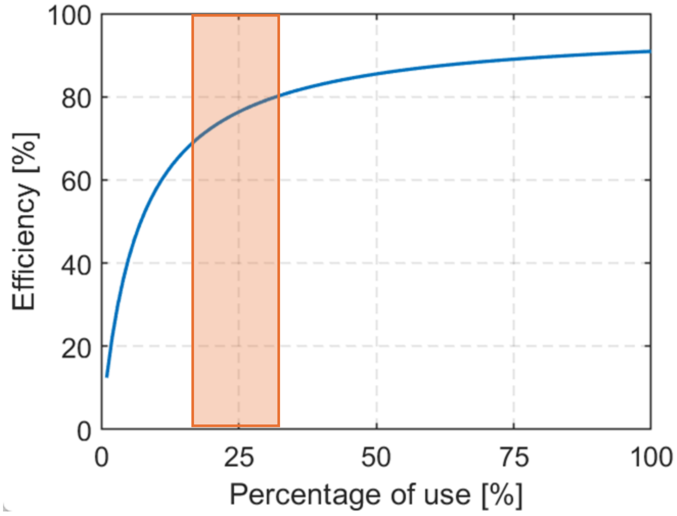

Figure 12: Theoretical performance curve for reciprocating compressor machine. COMHPTES operating points relative to 12-cylinder machine are shown in orange.

Figure 13 shows how the COP is affected by the mechanical efficiency. It is important to consider the real power requirements of a heat pump application and how this translates to machine utilisation. This will be an important factor to consider as part of the techno-economic analysis, and with most industrial processes requiring more than 500 kW of heating, it is likely that machine utilisation will be higher resulting in higher mechanical efficiency and thus higher COP.

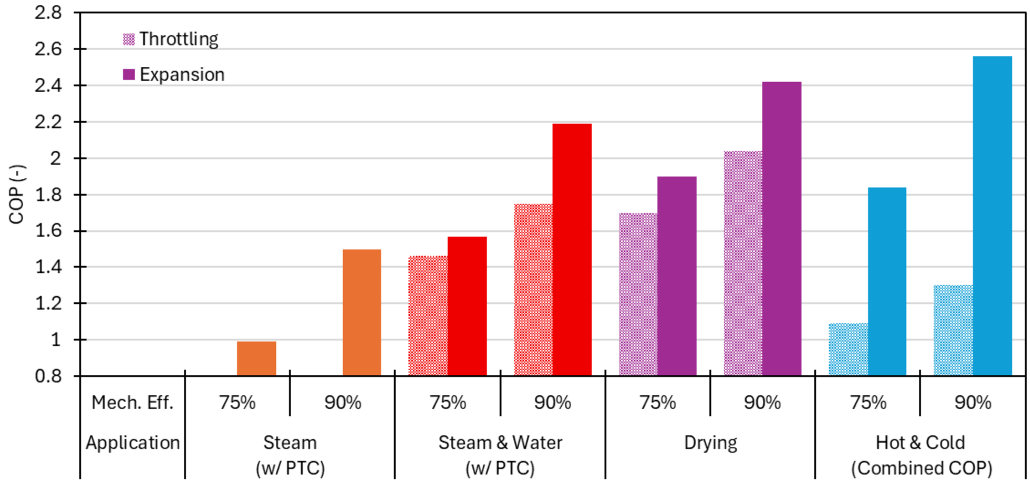

Figure 13: COP of heat pump cycles for base cases with and without expansion, showing effect of mechanical efficiency at part load.

4.2 Pinch Temperature

The minimum allowable pinch temperature directly impacts the size of the heat exchanger. In order to achieve very small pinch temperatures for example of 5 K the heat exchanger would need to be significantly larger than if a pinch temperature of 10 K was used. However, increasing the minimum pinch temperature, has a direct impact on the COP of the heat pump cycle, see Figure 14. As the pinch temperature in the COMHPTES system also represents the heat transfer fluid circuits on both the hot and cold side, the pinch temperature will need to be higher than the commonly accepted minimum of ~5 K for plate heat exchangers.

It should be noted that at pinch temperatures above 20 K, there is a notable loss in performance for the Hot & Cold cycle, with the cycle becoming unfeasible above a pinch temperature of 25 K, due to pressure ratio limitations and the saturation pressure of CO_2 at the cold side.

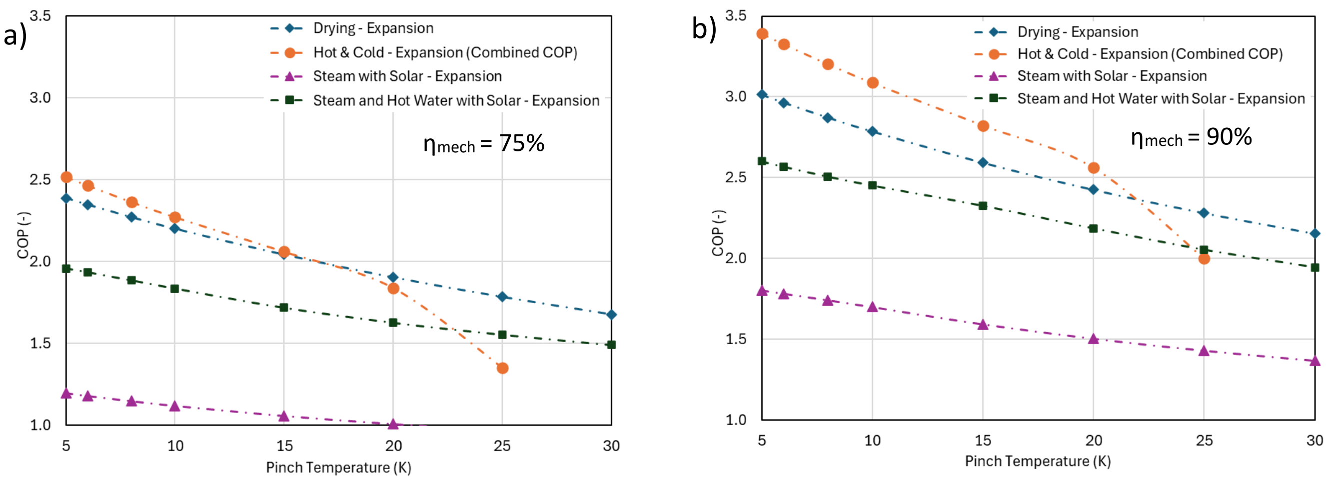

Figure 14: The effect of minimum pinch temperature in the source and sink on the COP of the base-case cycles at mechanical efficiency = a) 75% (test rig conditions) and b) 90% (full power conditions). Assumes 1% pressure drop.

4.3 Heat Exchanger Pressure Drop

As with pinch temperature, defining a smaller heat exchanger pressure drop is likely to increase the cost of the unit. Similarly, a higher pressure drop will also negatively impact the performance of the heat pump. This is illustrated for the drying example in Figure 15.

Pressure drop has a much smaller effect on the COP than other heat exchanger design parameters such as pressure and minimum approach temperature. Although it is still desirable to minimise the pressure drop in order to get the best possible performance, even pressure drops as high as 10% of the inlet pressure should not greatly impact the performance of the test rig (20 bar for HP and 4.1 bar for LP). The impact of pressure drop can be further analysed as part of the detailed techno-economic analysis.

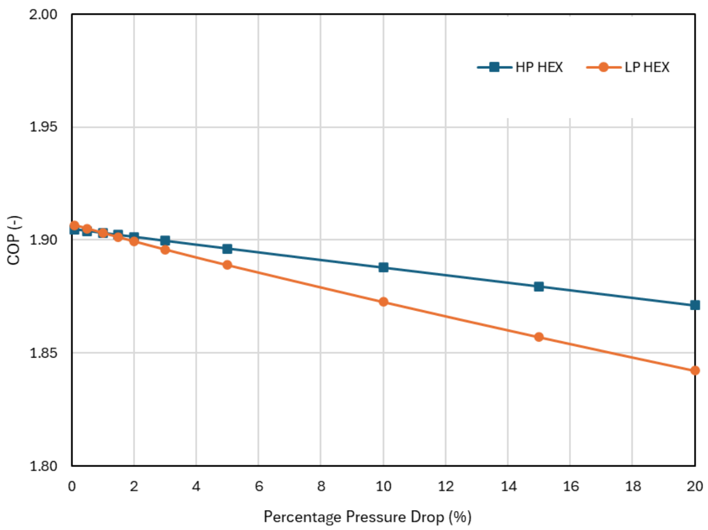

Figure 15: COP of Drying case study for changing pressure drop. The pressure drop of the other heat exchanger is kept constant at 1%. T_{pinch} = 20 K.

4.4 Heat Exchanger Design Pressure

The design pressure of the heat exchanger is defined by the maximum pressure of the working fluid, typically, 10% greater than the maximum operating pressure. Higher pressure heat exchangers are likely to be higher cost than those operating at lower pressure.

When optimising for the COP of a transcritical CO_2 heat pump, higher pressures are preferred as it improves power density. As per Table 6, SynchroStor's machine is limited to a maximum low pressure of 50 bara and a maximum high pressure of 200 bara. Plate heat exchangers up to a design pressure of 50 bar are readily available off the shelf from various vendors[5], [6] with units up to about 140 bar available for temperatures up to 150°C[6]. However, for higher pressures and temperatures, the solutions are typically more custom or require special designs, significantly increasing the cost and reducing availability.

Figure 16 shows the impact that the high pressure limit has on the COP of the heat pump cycle, with lower limits significantly reducing the efficiency. It should also be noted, that limiting the discharge pressure causes a reduction in the suction pressure (in order to achieve the same temperature lift) consequently, the density of the fluid at the compressor inlet is lower, reducing the mass flow per cylinder. In order to achieve the target power of 100 kWₑ, a greater number of cylinders is required to meet the mass flow requirements. In a real system, this would increase the cost of the heat pump, for the purpose of a demonstration rig, the power may become limited by the number of cylinders installed.

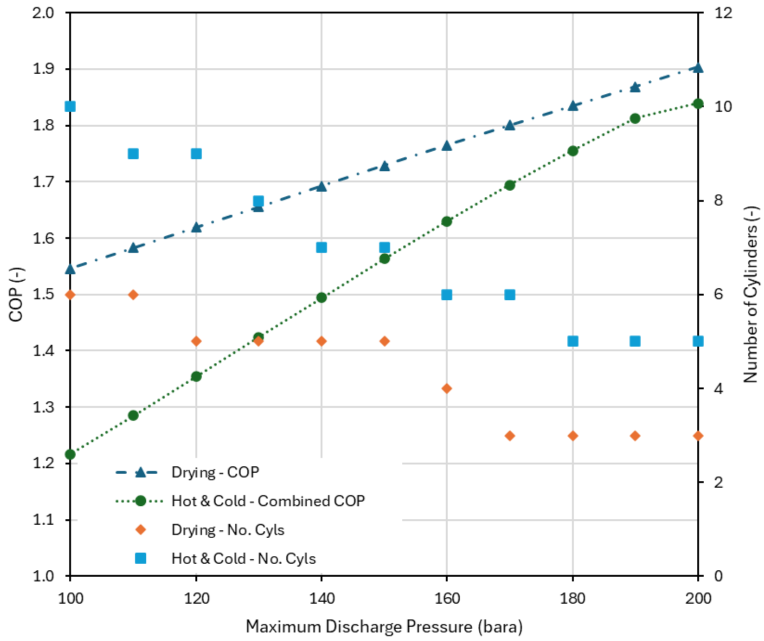

Figure 16: Effect of discharge pressure limit on heat pump COP and total number of cylinders (at 100 kW_e), for a heat pump cycle with expansion for Drying and Hot & Cold. T_{pinch} = 20 K, Pressure drop = 1%.

As SynchroStor's machine is designed for pressure up to 200 bar it is desirable to try to meet this target and maximise the effectiveness of the heat pump for the demonstration project. The practicality of this will need to be assessed based on heat exchanger availability and cost, as price break points may exist in some suppliers' catalogues.

4.5 Recuperated Cycles

A recuperator is only effective in cycles that have a high temperature coming out of the heat sink and a relatively low source temperature. This is the case for cycles generating steam. However, for the two other cycles considered in this report, the recuperator does not significantly improve the performance.

In the case of the drying cycle, the recuperator is not useful at all as the inlet of the air into the heat sink is cooler than the heat source air, leaving no heat to be exchanged between the high pressure and low pressure side. For the hot & cold application, the optimal point converges towards the cycle without a recuperator. This is observed because the cycle is driven by the requirement to also produce cold which means that the evaporation pressure must be the same, thus there is no advantage in higher compressor inlet temperature as the pressure ratio is driven by the cold side rather than the hot.

Including a recuperator is unlikely to yield significant benefits to the COMHPTES demonstration project. Furthermore, the solar collector provides similar, and, in some cases, better performance compared to a recuperated cycle. Trade-offs and advantages of including a recuperator are likely to differ by application and should be considered as part of the detailed techno-economic analysis.

4.6 Flash Tank

The present analysis does not include a flash tank. The purpose of the vessel would be to separate vapour from liquid following expansion. The liquid can then be routed to the evaporator which would operate more efficiently due to the single-phase inlet conditions, Figure 17 (a). This could potentially lead to a smaller and cheaper evaporator. However, with this in mind, most cycles considered for COMHPTES rely on a fairly high degree of superheat in order to maximise the usage of waste heat in the process. Consequently, only a portion of the heat source is used for evaporation with the remainder used for superheating. As such, using a flash tank with an evaporator only, then mixing saturated vapour would result in a significant loss in performance due to a lower compressor inlet temperature. Conversely, the vapour could be superheated in a heat exchanger separate to the evaporator (Figure 17 (b)) but this would again drive up the equipment cost.

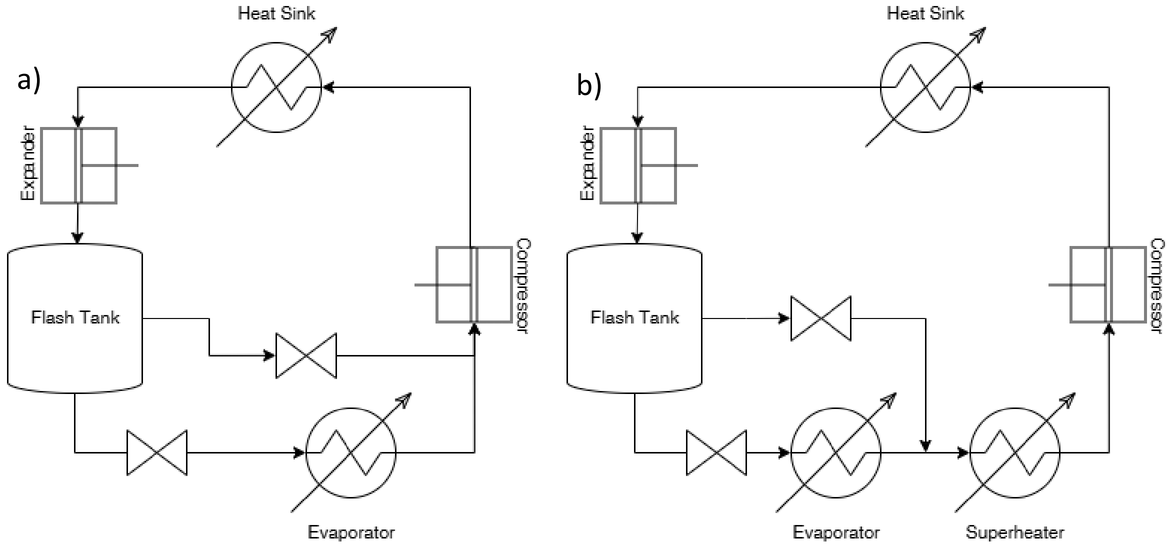

Figure 17: Heat pump diagram with flash tank showing a) evaporator only and b) evaporator and superheater.

For the purpose of the COMHPTES demonstrator it is likely to be more straightforward to not include a flash tank, although this may change with feedback from heat exchanger vendors and detailed design of the system. The techno-economic study may explore the implications of this decision.

4.7 Shaft Speed

Based on discussions with the KTH team it is understood that the lab where the demonstrator will be operated has two 160 kW bus bars. It is assumed that other equipment is connected to these, and the motor is powered by one of these, thus, the maximum peak load should not exceed 100 kW_e. Due to the limitation on drive power, imposed by the available lab supply, the number of cylinders is relatively low, especially for cycles with a high COP. SynchroStor's compressor expander machine is designed for up to 12 cylinders arranged radially around a central shaft. As such, very low cylinder counts do not make efficient use of the drive-train geometry and lead to higher losses. By reducing the shaft speed (by means of an inverter) the flowrate per cylinder can be decreased, resulting in more cylinders for the same electrical and thermal power. For the drying application, reducing the shaft speed from 1000 rpm to 500 rpm would increase the total cylinder count from 3 to 6. This may be beneficial for the COMHPTES project as it would improve the efficiency and make it easier to demonstrate the controllability of the heat pump system.

5. Conclusions and recommendations

The design of the COMHPTES demonstration test rig will need to be suitable for all operating conditions likely to be encountered throughout the testing program. Due to a lack of end-user applications at the time of writing, some generalised cases have been proposed to capture operating conditions that may be encountered across industry. Based on the present study, a set of design conditions may be derived for the pipework and heat exchangers forming the heat pump system. The heat transfer fluid interfacing with the heat pump is out of scope and will also need to be considered in the design of heat transfer equipment.

For the modelling in this report, a mechanical efficiency of 75% is used on account of a partially populated machine. This is due to the lab's constraint on the available electrical power supply. For a larger, real, system deployed in the field, one or more 12-cylinder machines would be used, with an estimated mechanical efficiency >90%, yielding higher COP. The lower efficiency value is used in this report to be representative of the conditions expected in the lab, but any techno-economic analysis should use a higher efficiency.

It is worth noting that the recuperated steam cycle is the only application that requires an IHX, as well as temperatures greater than 310°C. It is recommended to exclude that specific application from the testing phase and to consider it, instead, only as part of the techno-economic modelling exercise. Generally, steam is a difficult application for CO_2 heat pumps, however, the performance is greatly improved by including a solar thermal collector or including additional loads such as water heating on top of steam generation.

Table 13 and Table 14 show pressure and temperature design limit guidelines based on the considered case studies for the high pressure and low pressure equipment, respectively. The values presented are preliminary only and should be reviewed during detailed design to ensure conformance with relevant standards. It should be noted that the values below are for a base case of 10 K pinch temperature and 1% pressure drop in heat exchangers. If a different base case is used such as higher approach temperatures the boundaries are likely to change.

Table 13: Proposed design specification for high pressure equipment

Parameter	Unit	Value
Design Pressure	barG	220
Max. Operating Pressure	barG	199
Min. Operating Pressure	barG	0
Design Temperature (min / max)	°C	0 / 350
Operating Temperature (min / max)	°C	15 / 315

Table 14: Proposed design specification for low pressure equipment

Parameter	Unit	Value
Design Pressure	barG	55
Max. Operating Pressure	barG	49
Min. Operating Pressure	barG	0
Design Temperature (min / max)	°C	-29 / 165
Operating Temperature (min / max)	°C	-20 / 140

For the COMHPTES demonstration rig, it may be beneficial to install an expansion valve in parallel with the expander. This would allow for the benchmarking of the COMHPTES system against a typical heat pump. Furthermore, part of the techno-economic analysis could compare the cost and performance of a cycle with expansion versus one without. Some cycles, such as ones with a large glide and lower temperatures before expansion, would not recover much energy with expansion, e.g., drying.

In the present study, heat exchanger pinch temperature has been used as an analogue encompassing the minimum temperature difference in heat exchangers but also the buffer between the process fluid and heat pump working fluid due to the intermediate heat transfer and TES circuit. Whilst increasing the pinch temperature decreases the cost of the heat exchangers it also reduces the COP of the heat pump system. A value of 20 K has been used which yields a noticeable decrease on the COP of the heat pup compared to a more conventional value of 5-10 K. the higher pinch temperature is a result of including the

Heat exchanger pressure loss is less important in terms of heat pump performance, with pressure drops in the heat source having a greater impact. Heat exchanger design should aim to keep pressure loss below ~5% but this constraint can be relaxed if it leads to significantly cheaper equipment.

Limiting the design pressure of the system is another means by which system cost can be reduced, but it results in reductions in COP and does not fully utilise the design pressure and capabilities of SynchroStor's compressor/expander. Price breakpoints may exist in heat exchanger manufacturers' offering and this may justify limiting the design pressure of the system, similarly it may show that above a certain pressure there is little effect on price.

6. References

[1] Tetra Pak, *Dairy Processing Handbook*. 2025. [Online]. Available: https://dairyprocessinghandbook.tetrapak.com/chapter/primary-production-milk

[2] A. S. Mujumdar, *Handbook of Industrial Drying (4th Edition)*. Taylor & Francis, 2015.

[3] S. Madeddu and et al., 'The CO2 reduction potential for the European industry via direct electrification of heat supply (power-to-heat)', *Environ. Res. Lett.*, no. 15, 2020.

[4] I. H. Bell, J. Wronski, S. Quoilin, and V. Lemort, 'Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp (http://www.coolprop.org/)', *Ind. Eng. Chem. Res.*, vol. 53, no. 6, pp. 2498–2508, Feb. 2014, doi: 10.1021/ie4033999.

[5] SWEP, 'Product Range'. Accessed: Jun. 06, 2025. [Online]. Available: https://www.swepgroup.com/product/our-product-range.html

[6] Kelvion, 'Brazed Plate Heat Exchangers'. Accessed: Jun. 06, 2025. [Online]. Available: https://www.kelvion.com/products/plate-heat-exchangers/brazed-plate-heat-exchangers

Appendix

The conditions of the heat pump cycles for the base case are presented in this appendix.

Working Fluid:	CarbonDioxide										
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	5.19E+05	519	41.40	74.7	2108	-1.000	72	0.90	117.1	Wc
Comp	Hot HX	6.40E+05	640	200.00	231.1	2132	-1.000	232	0.90	290.7	Qh
Hot HX	Exp	3.17E+05	317	198.0	56.90	1325	-1.000	740	0.90	-19.2	We
Exp	Cold HX	2.94E+05	294	41.82	7.05	1334	0.368	269	0.90	203.1	Qc

Figure 18: Thermo cycle for drying with expansion with base case parameters.

Working Fluid:	CarbonDioxide										
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	5.24E+05	524	36.28	74.7	2142	-1.000	62	0.67	97.5	Wc
Comp	Hot HX	6.58E+05	658	200.00	245.2	2168	-1.000	222	0.67	245.4	Qh
Hot HX	Exp	2.92E+05	292	198.0	46.56	1249	-1.000	802	0.67	0.0	We
Exp	Cold HX	2.92E+05	292	36.65	1.91	1335	0.390	226	0.67	155.2	Qc

Figure 19: Thermo cycle for drying with throttling with base case parameters.

Working Fluid:	CarbonDioxide										
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	4.44E+05	444	25.22	-5.4	1941	-1.000	64	1.14	144.0	Wc
Comp	Hot HX	5.61E+05	561	200.00	174.4	1967	-1.000	288	1.14	204.2	Qh
Hot HX	Exp	3.82E+05	382	198.0	81.91	1515	-1.000	577	1.14	-46.5	We
Exp	Cold HX	3.38E+05	338	25.47	-11.37	1534	0.629	104	1.14	121.2	Qc

Figure 20: Thermo cycle for hot & cold with expansion with base case parameters.

Working Fluid:	CarbonDioxide										
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	4.50E+05	450	22.42	-5.33	1978	-1.000	54	0.70	97.5	Wc
Comp	Hot HX	5.79E+05	579	200.00	186.8	2007	-1.000	273	0.70	141.0	Qh
Hot HX	Exp	3.77E+05	377	198.0	79.86	1500	-1.000	590	0.70	0.0	We
Exp	Cold HX	3.77E+05	377	22.64	-15.40	1695	0.781	76	0.70	50.8	Qc

Figure 21: Thermo cycle for hot & cold with throttling with base case parameters.

Working Fluid:	CarbonDioxide										
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	5.85E+05	585	50.00	139.8	2250	-1.000	70	1.37	191.3	Wc
Comp	Hot HX	7.15E+05	715	200.00	289.9	2273	-1.000	197	1.37	188.9	Qh
Hot HX	Exp	5.77E+05	577	198.0	184.55	2003	-1.000	273	1.37	-93.8	We
Exp	Cold HX	5.03E+05	503	50.50	67.10	2027	-1.000	96	1.37	113.4	Qc

Figure 22: Thermo cycle for steam with solar with expansion with base case parameters.

Working Fluid:	CarbonDioxide											
Thermodynamic Cycle		H		P	T	S	Q	D	w	W/Q	
Outlet	Inlet	J/kg	kJ/kg	bara	°C	J/kg/K	-	kg/m^3	kg/s	kW	
Cold HX	Comp	5.85E+05	585	50.00	139.8	2250	-1.000	70	1.26	152.7	Wc
Comp	Hot HX 1	6.98E+05	698	170.54	271.0	2271	-1.000	176	1.26	130.0	Qh
Hot HX 1	Hot HX2	5.94E+05	594	168.8	189.78	2067	-1.000	224	1.26	130.0	Qh2
Hot HX2	Exp	4.91E+05	491	167.1	121.33	1826	-1.000	319	1.26	-54.70	We
Exp	Cold HX	4.44E+05	444	50.50	26.39	1844	-1.000	131	1.26	177.9	Qc

Figure 23: Thermo cycle for steam and hot water with solar with expansion with base case parameters.